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FLUIDIC SERVOACTUATORS FOR THREE-AXIS FLUIDIC STABILITY AUGMENTATION SYSTEM

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By

T. S. Honda

September 1970

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DAAJ02-70-C-0007

SPECIALTY FLUIDICS OPERATION, NBDO
RESEARCH AND DEVELOPMENT CENTER
GENERAL ELECTRIC COMPANY
SCHENECTADY, NEW YORK



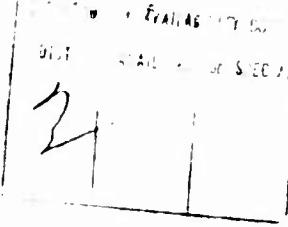
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Contract DAAJ02-70-C-0007
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September 1970

**FLUIDIC SERVOACTUATORS FOR THREE-AXIS
FLUIDIC STABILITY AUGMENTATION SYSTEM**

Final Report

By

T. S. Honda

Prepared by

**Specialty Fluidics Operation, NBDO
Research and Development Center
General Electric Company
Schenectady, New York**

for

**U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA**

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ABSTRACT

This report covers the developmental work accomplished on a program to furnish three fluidic servoactuators for a three-axis fluidic stability augmentation system (FSAS). The servoactuators, designed to the specification for a helicopter flight control system, are classified as "experimental" but were designed to meet safety-of-flight requirements since they were intended for flight test in a hydrofluidic SAS.

A servoactuator controlled by a two-stage fluidic servovalve was analyzed and designed. The servoactuator hardware was fabricated and bench tested. The servoactuator bench testing, using a load simulator, showed that the fluidic servoactuator meets all of the basic acceptance test requirements. The salient features are maximum force output of 100 lb, position frequency response flat to 7 Hz when operating with MIL-H-5606 oil supplied at 1000 psig, and a quiescent flow of 0.6 gpm.

FOREWORD

This report was prepared by the Specialty Fluidics Operation, General Electric Company, as part of U. S. Army Contract DAAJ02-70-C-0007, "Fluidic Servoactuators for Three-Axis Fluidic Stability Augmentation System" (Task IF162203A14186). The work was administered under the direction of the Aeromechanics Division, U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia, with Mr. George Fosdick as the US AAVLADS Technical Representative. The work was performed between 1 November 1969 and 20 June 1970, with Mr. T. S. Honda as Project Leader.

The author acknowledges the cooperation and support given by Mr. George Baltus of the Hydraulic Research and Manufacturing Company, who provided machining and fabrication services for the servoactuators used in this program.

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LIST OF SYMBOLS

A ₁	actuator piston small area - in. ²
A ₂	actuator piston large area - in. ²
A _B	bellows effective area - in. ²
A _N	nozzle projected area - in. ²
AR	amplitude ratio
B	fluid bulk modulus - psi
C'	fluidic capacitance - in. ³ /psi
C ₁	upstream vortex valve control pressure - psig
C ₂	downstream vortex valve control pressure - psig
CP ₁	fluidic input pressure to capsule plus side - psig
CP ₂	fluidic input pressure to capsule minus side - psig
CPS	cycles per second
cis	cubic inch per second
D	servoactuator damping coefficient - lb-sec/in.
F	force - lb
Hz	cycles per second
K _B	bellows spring gradient - lb/in.
K _T	suspension torsional spring rate - in. -lb/rad
K _Z	feedback wire spring gradient - lb/in.

K_y	feedback link ratio - in. / in.
K_θ	net spring rate - in. -lb/rad
M	load inertia - lb-sec ² /in.
P_1	valve output pressure to A_1 - psig
P_2	valve output pressure to A_2 - psig
P_3	servo pressure to A_1 - psig
P_4	servo pressure to A_2 - psig
P_{R1}	hydraulic return pressure from servoactuator - psig
P_{R2}	hydraulic return pressure from signal generator - psig
P_{S1}	hydraulic supply pressure to fluidic servoactuator - psig
P_{S2}	hydraulic supply pressure to signal generator - psig
psi	pounds per square inch
Q	flow rate - cis
R	denotes restrictor
r_B	capsule to pivot radius - in.
r_N	nozzle to pivot radius - in.
r_Z	feedback to pivot radius - in.
S	Laplace operator
T_ϵ	torque error - lb-in.
V	actuator entrained volume - in. ³
X	flapper displacement at nozzle - in.
Y	actuator piston displacement - in.
Z	feedback spring displacement - in.

∂ partial derivative
 Δ incremental change
 θ phase shift - degrees
 ω frequency - rad/sec

INTRODUCTION

This report summarizes the work performed under U.S. Army Contract DAAJ02-70-C-0007. The program involved the design, fabrication, and acceptance test evaluation of three prototype fluidic vortex valve servo-actuators for use in a helicopter hydrofluidic stability-augmentation-system flight test program. The specified static and dynamic performance requirements were met with a quiescent flow demand of under 0.6 gpm, which is significantly under the specified flow limitation of 0.8 gpm.

The servoactuator developed in the program is shown in Figure 1. It accepts a hydrofluidic (may also be pneumatic-fluidic) input signal from the FSAS computer and produces a proportional output displacement. The servoactuator functions as an extensible link in the primary flight control system to position the helicopter rotor pitch-control boost actuator.

The servoactuator is classed as a fluidic component in that the input torque motor is fluidic and a vortex amplifier circuit is used as the primary flow control stage in place of the conventional spool valve. This approach was developed to make an all-fluidic stability-augmentation system feasible. It provides a high-performance flight control servoactuator that is highly reliable and at a potentially lower cost than for conventional types.

The significant design operating conditions and performance requirements are listed in Table I.

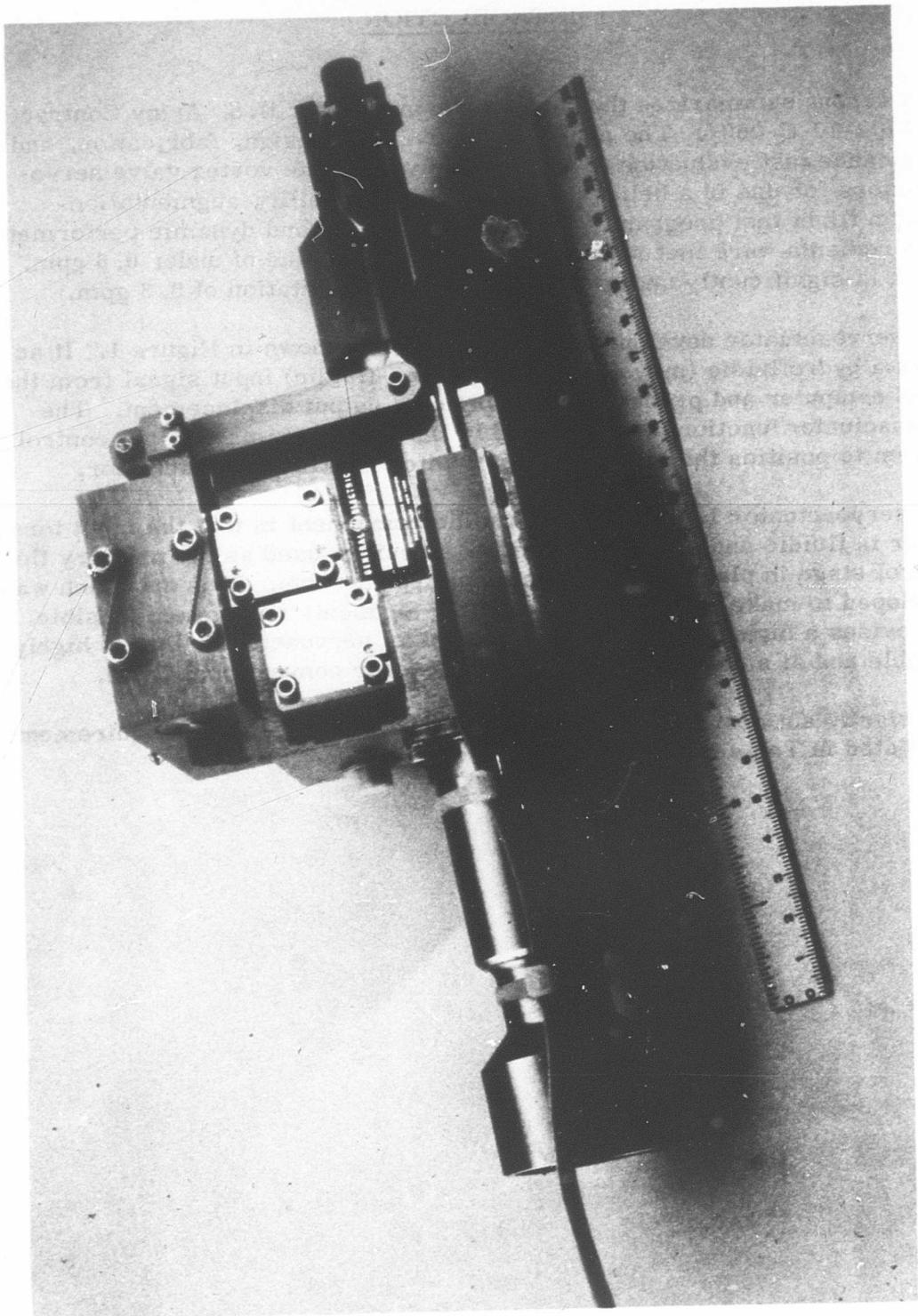


Figure 1. Fluidic Servoactuator.

TABLE I. DESIGN PARAMETERS

Service fluid specification	MIL-H-5606
Servoactuator supply pressure	1000 \pm 100 psig
Servoactuator return pressure	60 \pm 50 psig
Maximum flow per servo	0.8 gpm
Fluid temperature	100° + 20° F - 10° F
Fluidic input signal	\pm 4 psid
Input quiescent pressure	1306 psig
Servoactuator stroke	0.86 to 0.94 inch
Servoactuator gain	0.112 in./psid
Servoactuator stall force	100 lb
Inertia load	1.87 slugs
Friction load	5 lb
Response bandwidth	7-10 Hz
Servoactuator threshold	1% of full input signal

FLUIDIC SERVOACTUATOR DESIGN SPECIFICATIONS

The structural, static, and dynamic performance specifications for the fluidic servoactuator were defined to meet the requirements of a helicopter flight control system. A brief description of the servoactuator function and definition of the major requirements of the power supply, as well as input and output interfaces, are presented in this section.

SERVOACTUATOR FUNCTION

The servoactuators are installed as an extensible link in each of the three axes of the primary flight control system as shown in Figure 2. Their function is to drive the surface control system boost actuator pilot valve, thereby applying control inputs to damp out gust disturbances imposed on the aircraft. The servoactuator is a "position servo" whose output shaft position is proportional to the applied fluidic input signal pressure. In the absence of hydraulic pressure, the servoactuator ram is centered and locked such that the servoactuator functions as a rigid link in the primary flight control system, thereby providing normal helicopter control without the stability augmentation feature.

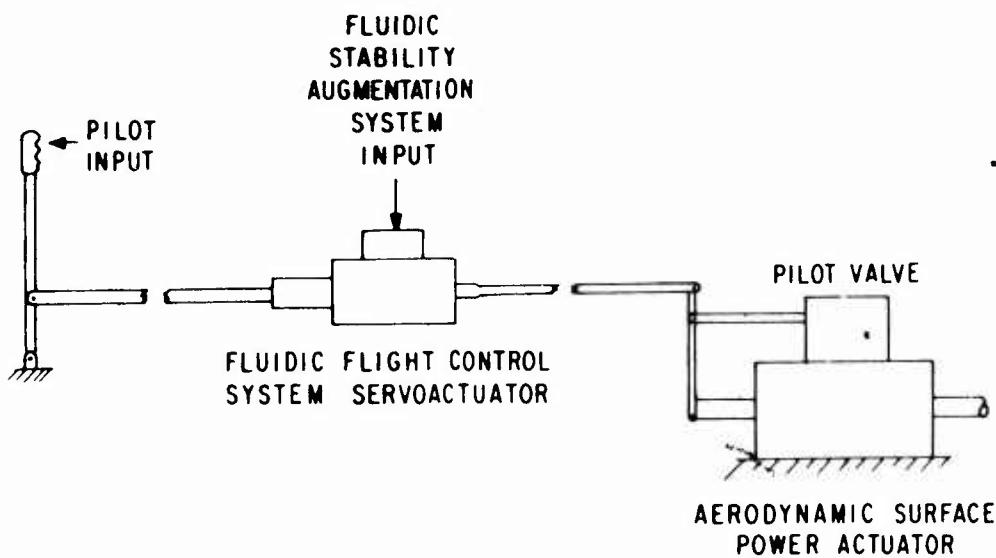


Figure 2. Primary Flight Control System Schematic.

HYDRAULIC POWER SUPPLY SYSTEM SPECIFIED

The relationship of the servoactuators in the hydraulic power system is shown in Figure 3. The servoactuators operate downstream of the fluidic stability-

augmentation-system sensors and computer, which are supplied from a 1500-psig hydraulic pressure source. The servoactuator supply pressure is regulated at 1000 psig, and the drain is to a 60-psig hydraulic return. The maximum flow available for operation of the servoactuators is limited to that demanded by the sensors and computing circuitry. Thus, the quiescent flow requirement for each actuator must be less than 0.8 gpm.

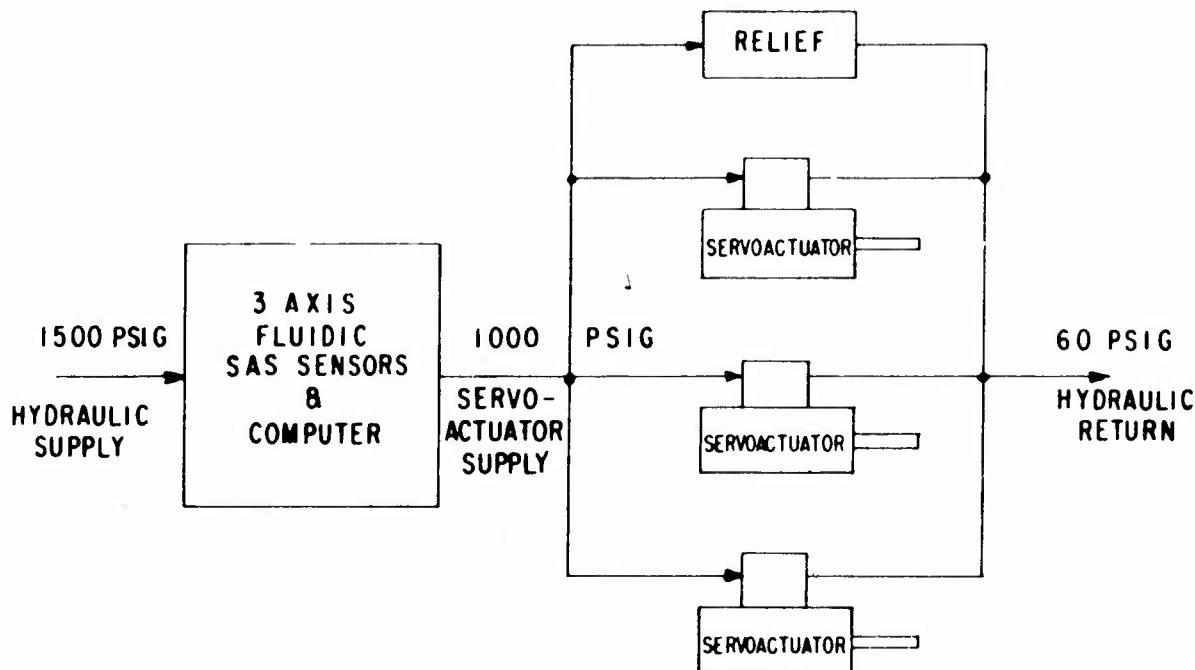


Figure 3. Servoactuator Power Supply Schematic.

The hydraulic fluid must meet the requirements of specification MIL-H-5606 at a temperature ranging from 60° to 185°F.

FLUIDIC INPUT INTERFACE REQUIREMENTS

The input to the servoactuator interfaces with the fluidic computing circuitry. The rated amplitude of the hydraulic-fluidic input signal is ± 4 psid. The nominal quiescent pressure level of the input signal is 1306 psig.

The fluidic torque motor to which the input is applied is required to have an effective fluidic capacitance of not more than 0.0005 in³/psi per side when operating in the servoactuator at blocked and rated conditions. This capacitance is defined as

$$C' = \frac{(A_B)^2}{K_\theta}$$

where A_B is the bellows or diaphragm effective area and K_θ is the net spring gradient of the torque motor.

SERVOACTUATOR OUTPUT REQUIREMENTS

The servoactuator output is a linear shaft displacement of ± 0.45 inch. The static gain from input fluidic signal to output displacement required is 0.112 inch per psid. The output stall force capability must be at least 100 lb.

The mass load reflected to the servoactuator output is 1.87 slugs, and the friction load is 5 lb. The servoactuator with full load is required to have a frequency response equivalent to a quadratic system with $\omega_o = 62.8$ rad/sec and with a damping ratio of 0.5 to 1.0.

The servoactuator piston centerlock mechanism is to be fully unlocked within 0.5 second after application of rated supply pressure. The centerlock is to begin to release at 125 psig and be completely released at a pressure not exceeding 300 psig. The servoactuator null positions in the centerlocked and active conditions are to be within 2% of full stroke at no load.

The servoactuator is required to centerlock from a fully extended or retracted position in 1 second with a 40-lb opposing load. The servoactuator and centerlock must also be capable of holding a 1500-lb tensile or compressive load when the servoactuator is deactivated.

FLUIDIC SERVOACTUATOR DESIGN

The servoactuator design, shown schematically in Figure 4, consists of a two-stage fluidic servovalve and linear hydraulic actuator driven by a fluidic torque motor. An actuator piston centering and locking mechanism is provided for safe operation upon the loss of hydraulic power. A mechanical feedback linkage is included to stabilize the actuator piston position. The servoactuator is fitted with a center-tapped potentiometer to provide electrical readout of proportional actuator position during flight tests.

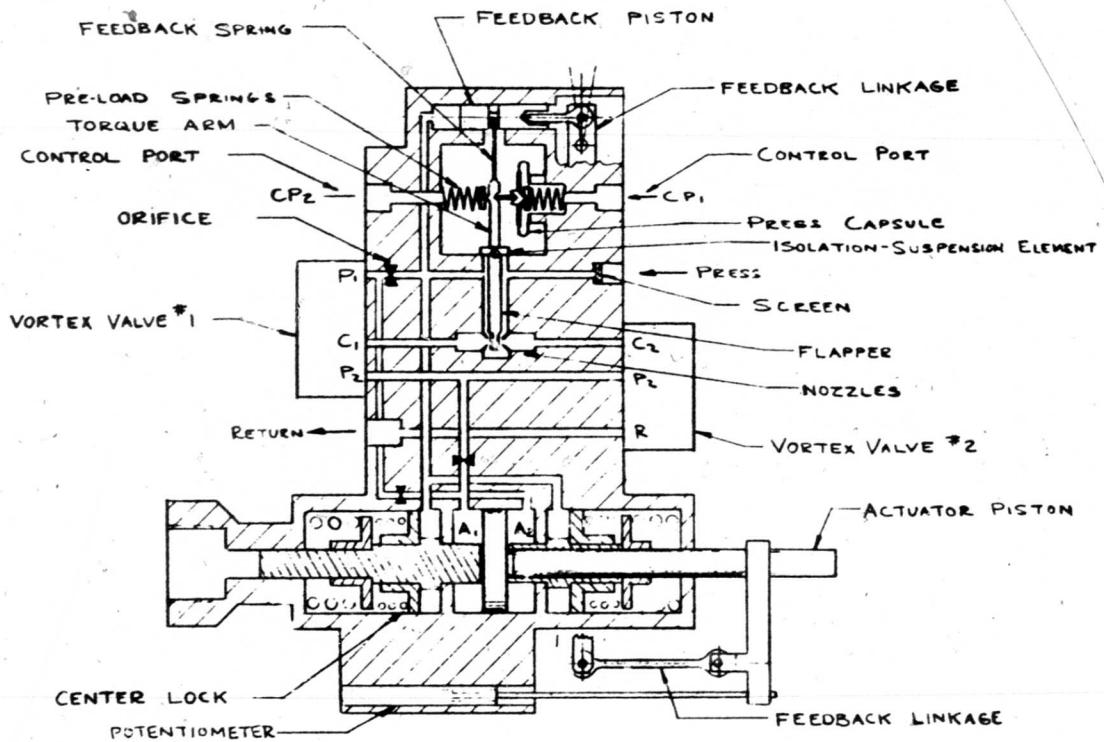


Figure 4. Fluidic Servoactuator Schematic.

The servoactuator outline as shown in Figure 5 conforms to the space envelope limits of a current-production Bell helicopter. The servoactuator body contains four ports to provide the interface with the two-fluidic input lines, the hydraulic supply line, and the return line. Details of the fluidic torque motor,

servovalve, and linear actuator design are presented below, and the dynamic analysis used to establish the required component and loop gains is presented in Appendix I.

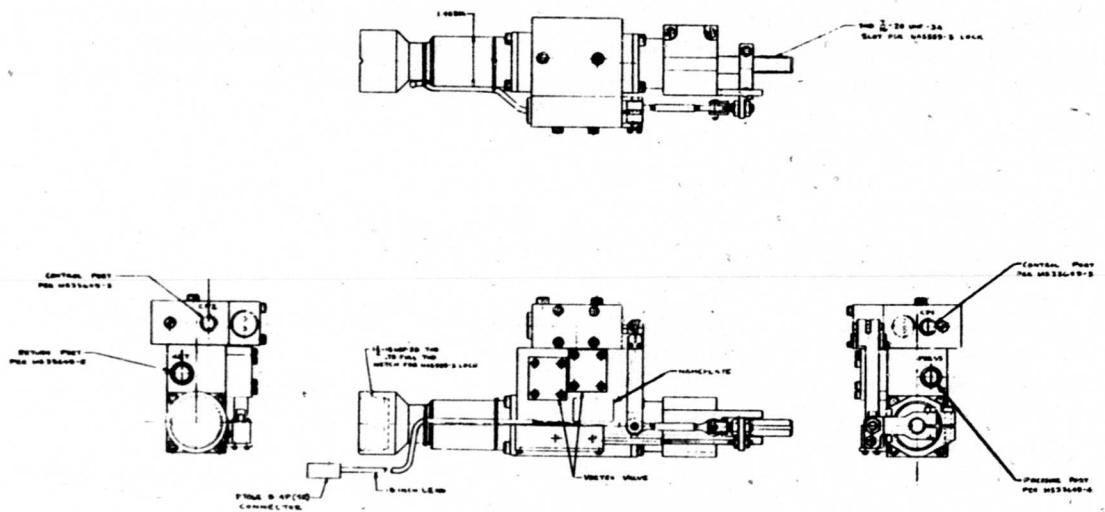


Figure 5. Fluidic Servoactuator Outline Drawing.

FLUIDIC TORQUE MOTOR

The fluidic input-pressure and the actuator output position-feedback signals are summed in the torque or force motor to produce a first-stage flapper displacement proportional to the position-error signal. The input differential-pressure is applied across a thin-wall metal capsule to produce a force on the flapper input arm. The actuator displacement is reduced through the feedback linkage and introduced into the torque motor (as another force into the flapper input arm) through a wire spring arrangement. The input arm and flapper pivot about a flexure suspension and hydraulic isolation member. The torque motor also includes coil springs to provide preload and null adjustment.

The capsule effective area, input arm-flapper link ratio, and spring rates (capsule, feedback wire, suspension, and bias adjust) are selected to provide flapper stability while meeting the torque motor capacitance and actuator threshold specifications. The resultant design values of the main parameters are:

Capsule effective area	$A_B = 0.286 \text{ in}^2$
Net spring rate	$K_\theta = 88.6 \text{ in-lb/rad}$
Capsule to pivot	$r_B = 0.568 \text{ in.}$
Nozzle to pivot	$r_N = 0.52 \text{ in.}$
Feedback spool to pivot	$r_Z = 1.673 \text{ in.}$
Capacitance	$\frac{r_B A_B^2}{K_\theta} = 0.0003 \text{ in}^3/\text{psi}$

HYDRAULIC FLUIDIC SERVOVALVE

The hydraulic fluidic servovalve consists of a positive pressure feedback flapper-nozzle first stage and a series arrangement of two vortex valves and a fixed orifice as the second stage. This arrangement, shown schematically in Figure 6, was utilized successfully in a prior AVLabs program under Contract DAAJ02-68-C-0093.*

*Honda, T. S., and Ralbovsky, F. S., Fluidic Vortex Valve Servoactuator Development, General Electric Co., USAAVLabs Technical Report 69-23, U.S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia. May 1969, AD859804.

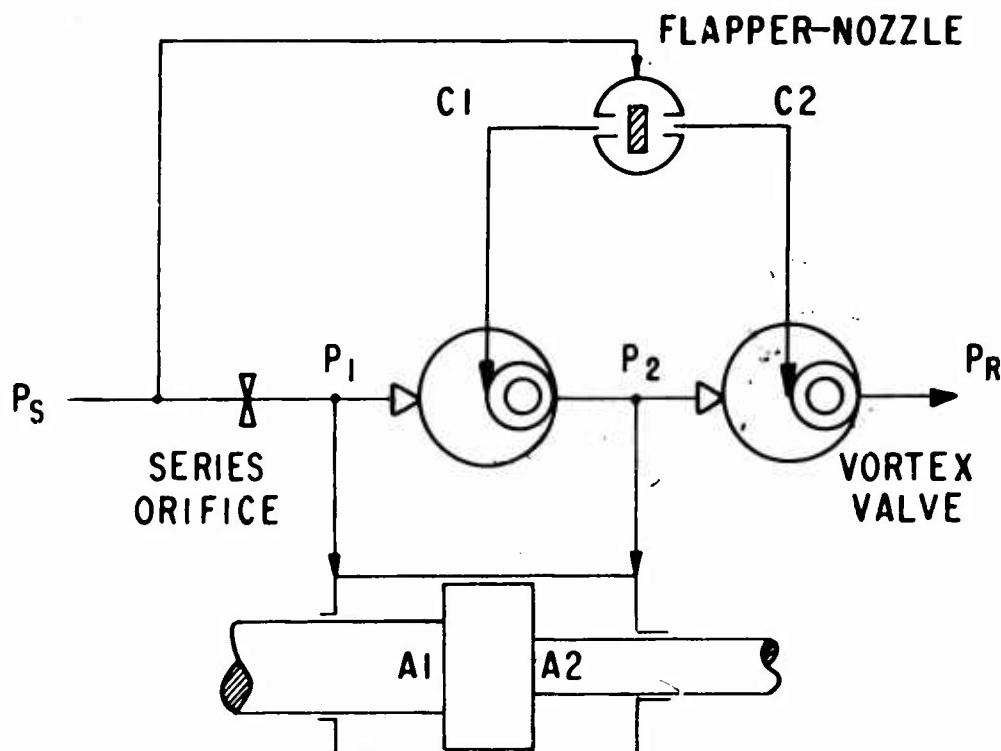
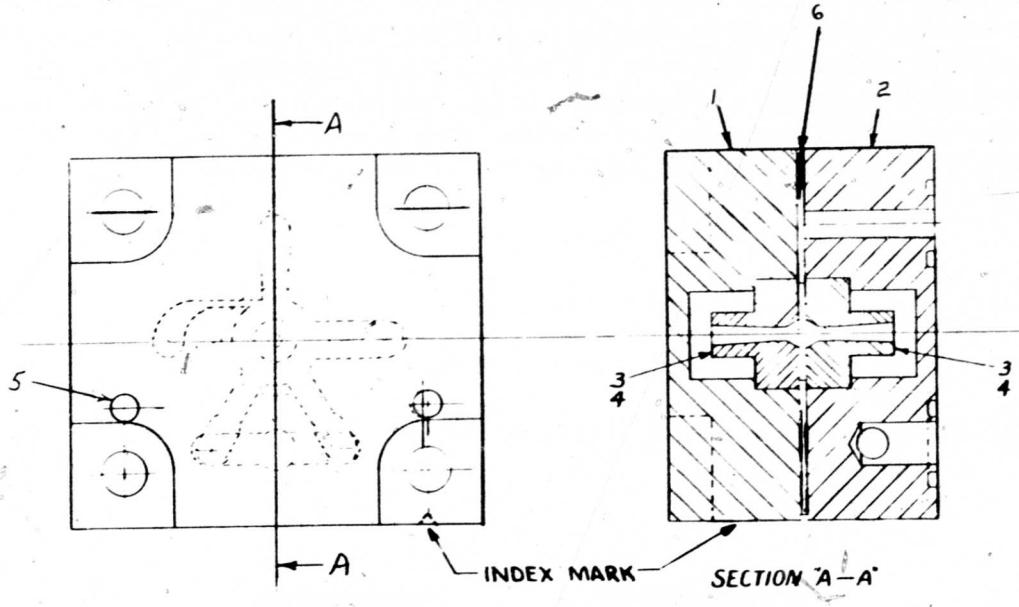


Figure 6. Vortex Valve Servoactuator Fluidic Circuit Diagram.

The flapper-nozzle, vortex valve, and orifice dimensions were established to provide the actuator force capability and loop gain at the specified supply pressure of 1000 psig and within the flow limitation of 0.8 gpm. The final design values selected for the pertinent parameters are:

Nozzle diameter	- 0.020 inch
Fixed orifice diameter	- 0.030 inch
Vortex valve spin chamber diameter	- 0.250 inch
Upstream vortex valve outlet diameter	- 0.031 inch
Downstream vortex valve outlet diameter	- 0.040 inch

Vortex valves are used in place of the conventional spool valve for the servoactuator second stage. The second stage thus has no moving parts. The vortex valves are designed as a subassembly which is mounted on the servoactuator body. Details of the vortex valve assembly are shown in Figure 7. It is a double-outlet, radial-inlet type which was designed specifically for



Part Identification

- | | |
|-----------------|-----------------|
| 1 cover | 4 outlet nozzle |
| 2 body | 5 dowel pin |
| 3 outlet nozzle | 6 laminations |

Figure 7. Vortex Valve Assembly.

this application. The central element of the vortex valve, which contains the control nozzle, spin chamber, and radial inlet passages, is comprised of a stack of chemically etched stainless steel laminations such as that shown in Figure 8. The laminations are diffusion bonded to provide zero leakage and structural integrity under high-pressure usage. The final stacking arrangement used starting from the cover end is:

one - VC 10

three - VC 9

two - VC 10

The lamination stack height is thus 0.020 inch.

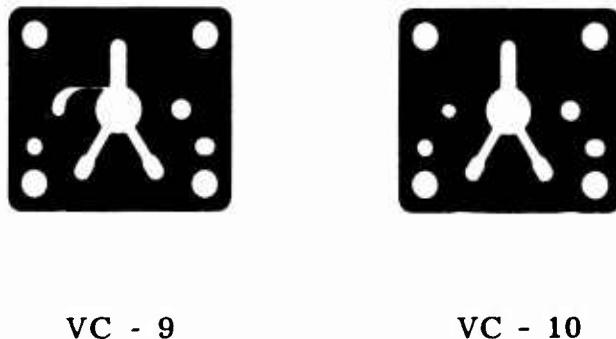


Figure 8. Vortex Valve Laminations.

HYDRAULIC LINEAR ACTUATOR

The servovalve drives a hydraulic linear actuator to provide the displacement and force requirements. The vortex valve circuitry utilized is optimized with the actuator piston area selection for minimum quiescent flow demand. Unequal piston areas are required for compatibility with the valving circuit; the optimum area ratio is 2:1.

In order to develop a stall force of 100 lb and provide for 15 lb internal and 5 lb external friction forces, the theoretical force required is 125 lb. The design values for the actuator cylinder pressures developed by the vortex valve circuitry under blocked load are:

	<u>Large Area Pressure (psig)</u>	<u>Small Area Pressure (psig)</u>
Extend	725	750
Retract	100	750

The resultant piston area requirements are 0.41 in² and 0.22 in². These areas in combination with the vortex valve design provides slew capability of 4 in. /sec against a 50-lb load.

The actuator centerlock mechanization is a conventional design modified to meet the specific requirements of this application.

RESULTS OF LABORATORY EVALUATION

Three fluidic servoactuators were manufactured for delivery to USAAVLABS. Prior to delivery, all servoactuators were subjected to an acceptance test program. Engineering tests on one servoactuator included evaluation at low temperature and in a vibration environment. Test data were also obtained for the vortex valves designed for this application. The results of these tests are summarized in this section.

ACCEPTANCE TEST

Evaluation of the fluidic servoactuators demonstrated that the structural design and basic performance conformed to design specifications. A comparison between test results and specified values of some of the significant performance parameters is shown in Table II. Details of the acceptance test procedure followed are included in Appendix II. The test facility used to conduct engineering and acceptance tests is described in Appendix III.

TABLE II. TEST RESULTS VERSUS DESIGN SPECIFICATIONS

	Specified	Actual
Quiescent flow (gpm)	0.8	0.5 to 0.6
Piston friction (lb)	<15	6 - 10
Centerlock time with 40 lb load (sec)	1.0	0.6 - 0.7
Threshold (%)	1	0.75 - 0.9
Pressure null shift (%/psi)	0.015	0.001
Stroke (in.)	0.86 - 0.94	0.865 - 0.910
Stall force (lb)	100	100 - 115
Static gain (in. /psi)	0.101 - 0.123	0.106 - 0.117
Dynamic response* (Hz)	10	6 - 7
Phase shift at 2 Hz (degrees)	25	12 - 17

*Bandwidth of an equivalent quadratic system; i. e., frequency where -6 db attenuation and 90° phase shift occur.

During the course of laboratory evaluation, it became apparent that an impedance matching problem between the fluidic torque motor and the driver amplifier existed. Since the fluidic amplifier is not an ideal pressure source, the amplifier output impedance coupled with the torque motor capacitance introduces a lag in the servoactuator feedback loop. The lag frequency occurs sufficiently close to the desired crossover frequency of 10 Hz to cause marginal stability of the servoactuator. Reduction of the servoactuator loop gain to provide a bandpass in the range of 6 to 7 Hz provided loop stability. A review of the stability-augmentation-system requirements indicates that this reduction in the servoactuator bandpass frequency could be tolerated. A typical closed-loop position response characteristic of the servoactuator is shown in Figure 9. The corresponding step response trace is shown in Figure 10.

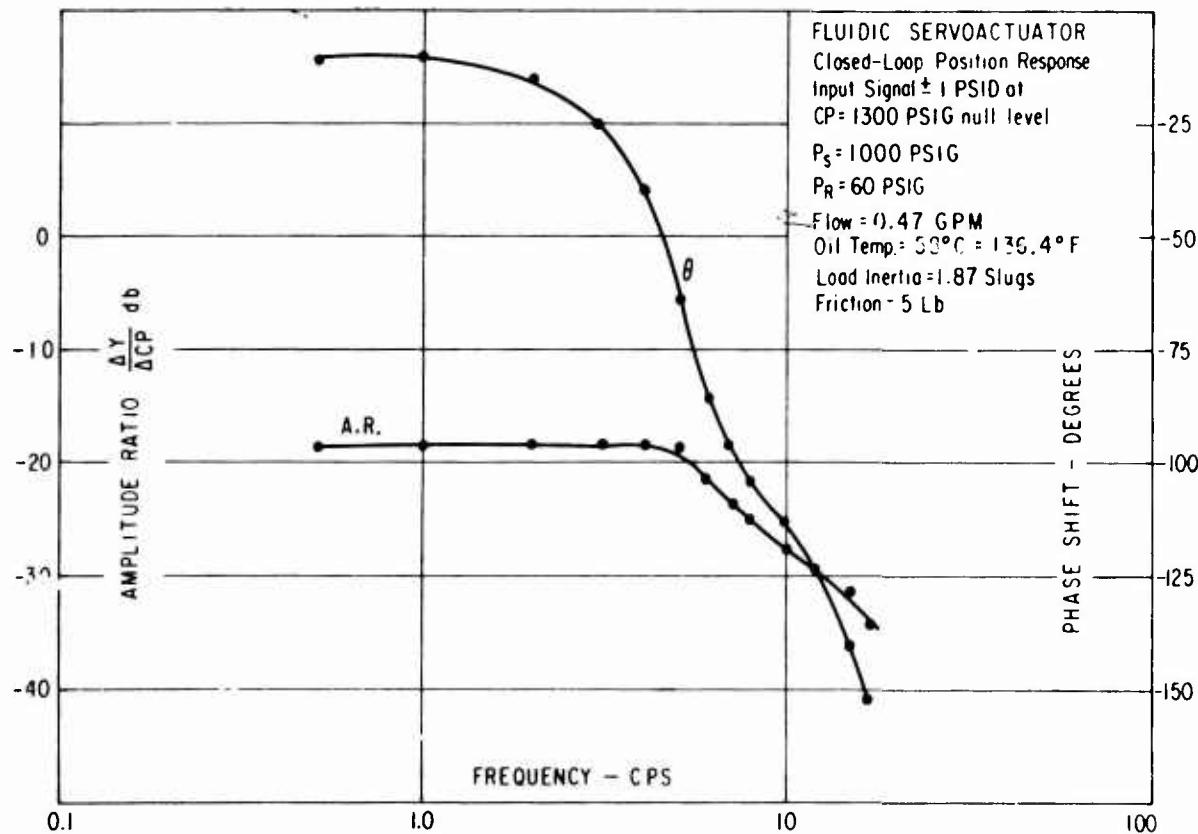
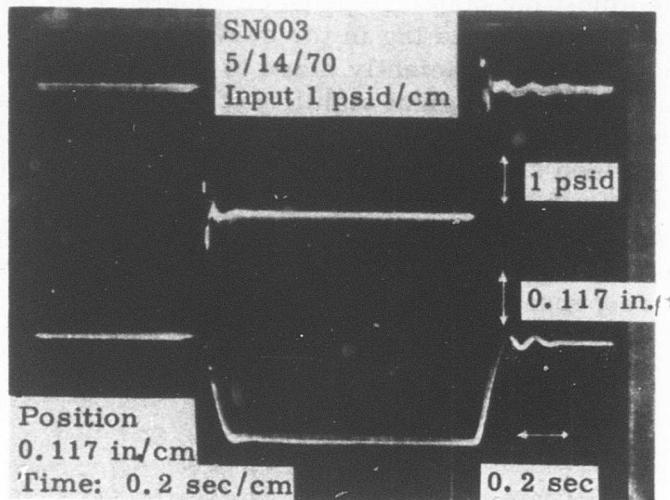


Figure 9. Fluidic Servoactuator Closed-Loop Position Response.



Inertia Load-1.87 Slugs
Friction Load-5 lb.

Figure 10. Fluidic Servoactuator Step Response.

The servoactuator static gain characteristic, traced on an X-Y recorder, as shown in Figure 11, shows excellent linearity. The small hysteresis of about 0.7% is primarily a function of the servoactuator friction and the servovalve pressure gain. The hysteresis, in effect, appears as the servoactuator threshold and is within the specified design requirements.

The servoactuator centerlock time was determined from observation of the supply pressure and servoactuator position traces on an oscilloscope. A photograph of a typical centerlock time trace is shown in Figure 12.

LOW-TEMPERATURE PERFORMANCE TEST

One of the program requirements was to obtain servoactuator performance data with fluid temperature down to 40°F. One servoactuator was operated over a fluid temperature range of 40° to 155°F to establish null shift, gain, and response changes.

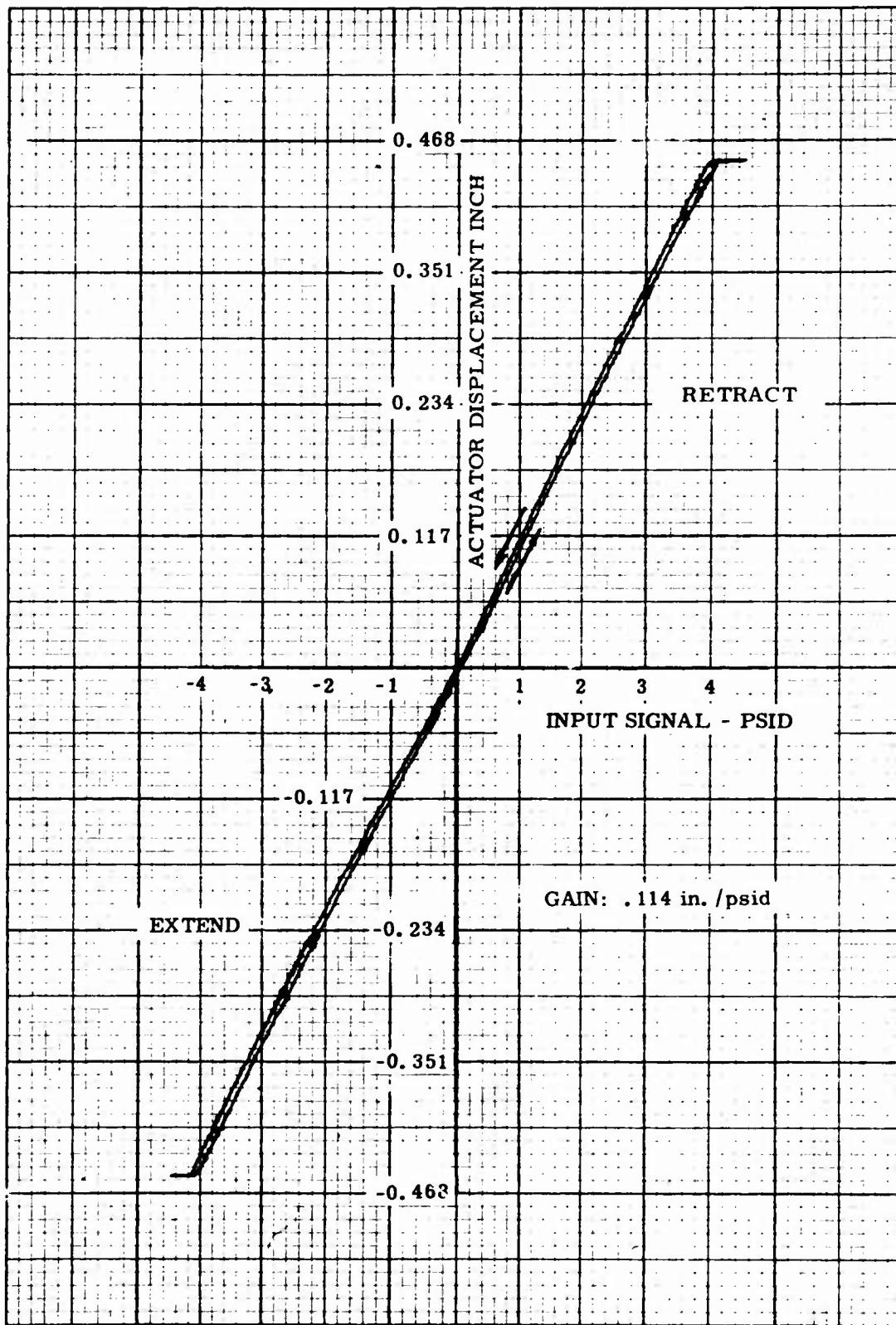
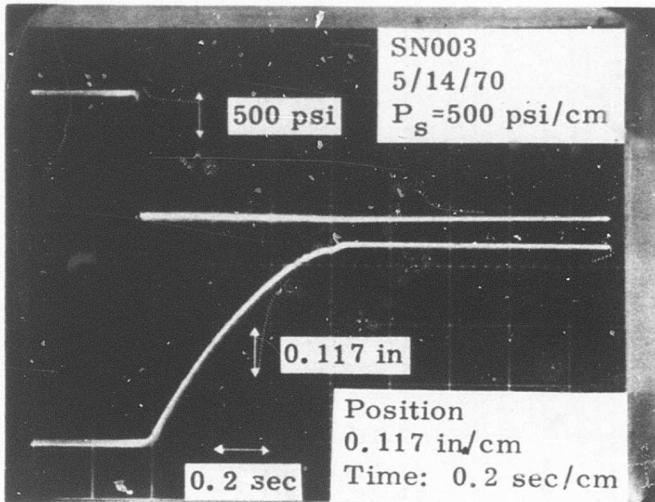


Figure 11. Fluidic Servoactuator Static Gain.



Inertia Load - 1.87 Slugs
 Force Load - 45 lb
 From Full Extended Position

Figure 12. Fluidic Servoactuator Centerlock Time.

The position gain-change over the fluid temperature range evaluated was negligible. The temperature null shift observed was less than 0.5%. The threshold in the temperature range of 40° to 155°F was within 1.25% of rated input signal. The change in response due to oil temperature is summarized in the Bode plots of Figure 13. These data show that the change in frequency response becomes insignificant above 115°F.

VIBRATION TEST

A vibration scan at the amplitudes and frequencies of MIL-STD-810B, Helicopter Test Curve B, was conducted on one servoactuator. The servoactuator was operating with a hydraulic supply pressure of 1000 psig and return pressure of 60 psig. The oil temperature ranged from 86° to 96°F during the tests. The input differential pressure was held at zero at a level of 1000 psig. A sinusoidal vibration cycling per the test envelope was

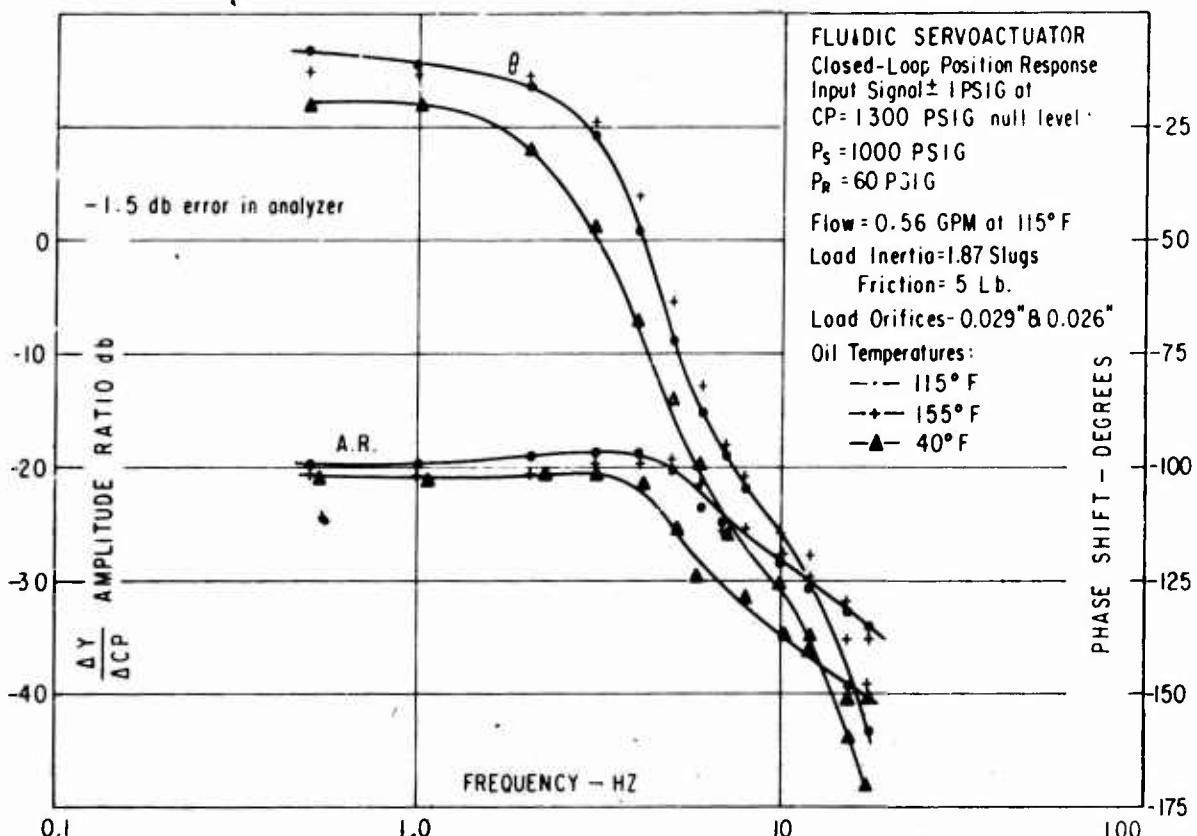


Figure 13. Fluidic Servoactuator Closed-Loop Position Response at Various Temperatures.

conducted at a rate sufficiently slow to allow adequate identification and evaluation of the resonant frequencies or functional phenomena. Sinusoidal vibration cycle of at least 15 minutes was performed in each axis.

The servoactuator exhibited no resonances at 5 and 10 Hz when subjected to the vibration environment. The effect of the vibration environment on null drift was noted as follows for each of the directions:

Axial - 0.22% of the total stroke

Vertical - 0.55% of the total stroke

Transverse - less than 0.1% of the total stroke

A resonance at an amplitude of approximately 0.009 inch (1% of full stroke) was observed in the frequency range from 55 to 80 Hz. This resonance was observed during tests in each axis.

The servoactuator's performance (gain, threshold, and response) after being subjected to the vibration environment was essentially unchanged from its performance prior to being subjected to the vibration environment.

VORTEX VALVE EVALUATION

Test data were obtained for the vortex valves designed for the servoactuator. The data obtained using MIL-H-5606 hydraulic fluid at the design operating temperature and in the pressure range around design null are presented in Figures 14 and 15.

Figure 14 shows the turndown characteristic of the upstream valve which is designed to operate with 750 psig supply pressure and 375 psig output pressure. Its turndown ratio (TDR) is approximately 5.9. Figure 15 shows the characteristics of the downstream valve, which is designed to operate with 375 psig supply pressure and drain into a 60 psig return pressure. Its TDR is 6.4. The valve gain established from the test data is 0.0675 in.³ per sec per psi.

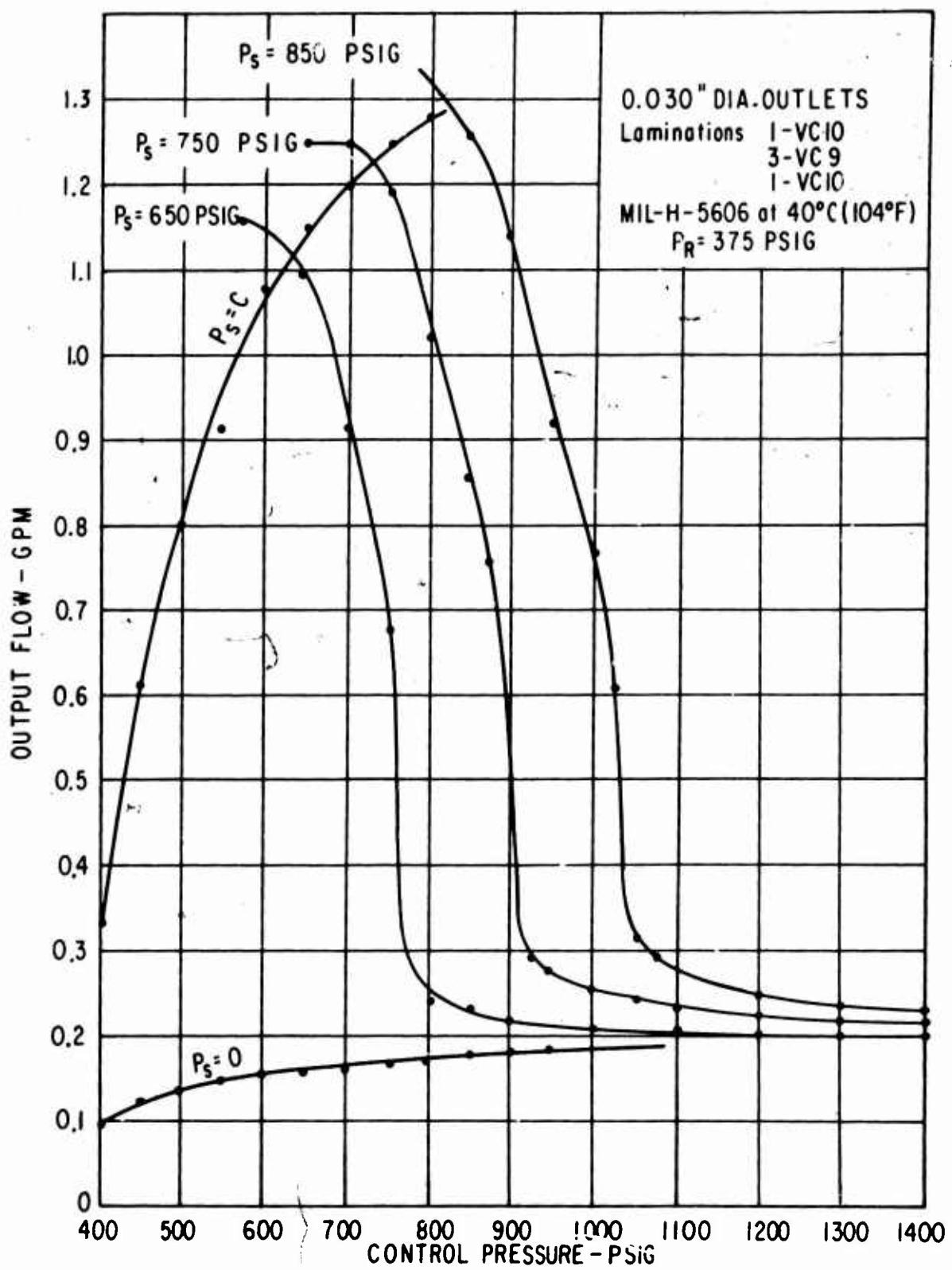


Figure 14. Vortex Valve No. 1 Test Data.

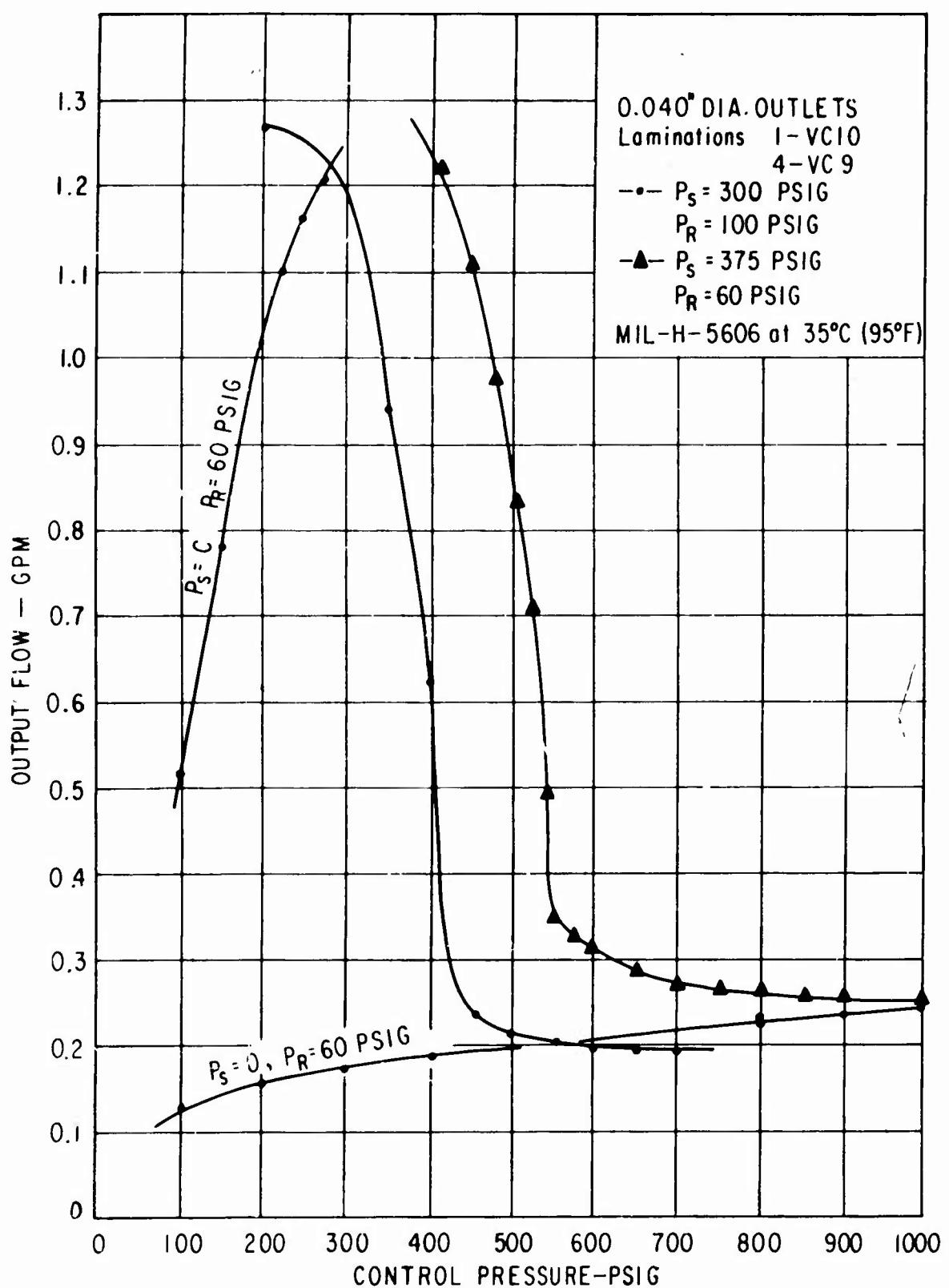


Figure 15. Vortex Valve No. 2 Test Data.

CONCLUSIONS

The data obtained and evaluation of test results from this program lead to the following conclusions:

1. The fluidic servoactuators designed and fabricated during this program meet the specified requirements of a helicopter stability-augmentation system.
2. The performance of the fluidic servoactuator is predictable by design analysis.
3. The manufacture of fluidic servoactuators with repeatable performance characteristics is practical.
4. The largest change in servoactuator response occurs with oil temperatures below 100° F. At oil temperatures above 100° F, the vortex valve Reynolds number is greater than 1000 and the valve gain becomes independent of viscosity.
5. The fluidic servoactuator developed can be controlled by a low-pressure pneumatic or hydraulic signal source as well as the intended high-pressure input level.
6. The servoactuator performs satisfactorily under the vibration environment of Helicopter Test Curve B, MIL-STD-810B.

RECOMMENDATIONS

The fluidic servoactuator dynamic response and stiffness were compromised to assure stable operation when driven by a hydrofluidic amplifier. Higher response can be achieved, however, by more detailed definition of the input impedance and by appropriate fluidic torque motor design. It is, therefore, recommended that the fluidic servoactuator input interface requirements be more thoroughly investigated and that the fluidic torque motor design be modified to realize the full response and stiffness capabilities of the actuator.

The fluidic servoactuator should be flight tested with the fluidic SAS, and refinements in design requirements should be identified on the basis of flight test evaluation.

APPENDIX I VORTEX VALVE SERVOACTUATOR ANALYSIS

Analysis of the vortex valve servoactuator shown schematically in Figure 16 is made to define the transfer function and the parameters pertinent to achieving the required static gain, response, and threshold. The transfer function is derived from the basic linearized equations for the flow, force, and displacement relationships in the system.

In order to simplify the analytical calculations, certain assumptions are made as follows:

1. For small changes about an initial condition, the dynamic relationships are linear.
2. The spring-mass dynamics in the fluidic torque motor contribute negligible phase shift.
3. There is no friction in the fluidic torque motor.
4. There is no leakage across the actuator piston or to ambient.
5. The torque motor hysteresis is negligible.
6. The input is a pressure signal with no associated flow dynamics.

The linearized equations describing the valve and actuator flow, force, and motion relationships are as follows.

Fluidic Torque Motor

A model of the torque summation on the torque motor flapper assembly is shown in Figure 17. The equation for the torque summation is

$$r_B A_B \Delta C_P + r_N A_N \Delta C - r_z K_z \Delta z = \frac{K_\theta}{r_N} \Delta X \quad (1)$$

where $K_\theta = r_B^2 K_B + K_T + r_z^2 K_z$ (2)

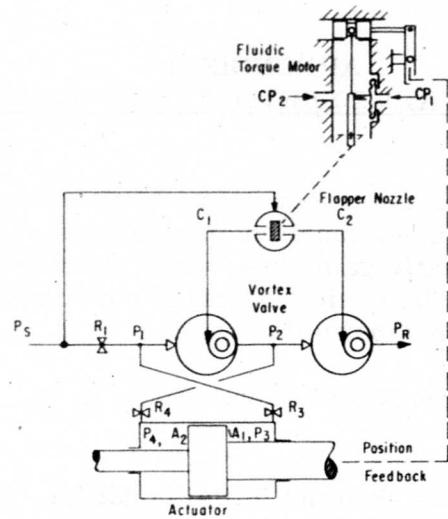


Figure 16. Fluidic Servoactuator, Fluidic Circuit Diagram.

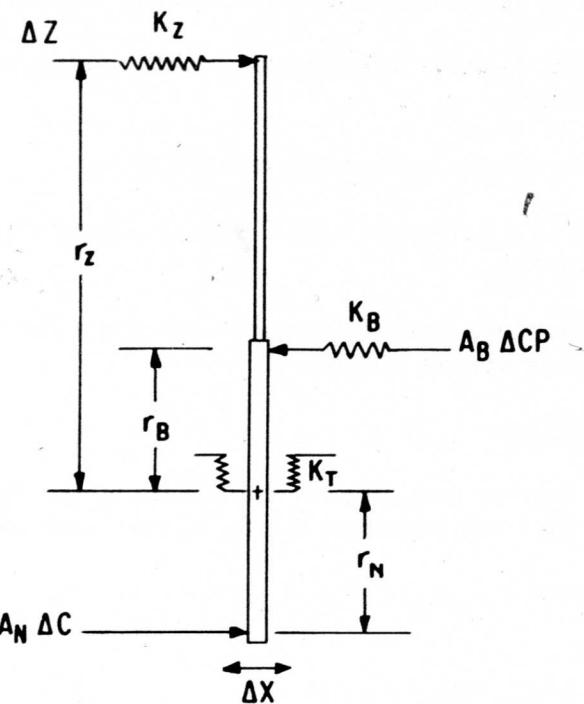


Figure 17. Flapper Torque Summation Model.

The spring-mass resonance of the flapper assembly is estimated to be in the order of 2000 rad/sec, and its effect on the system dynamics is neglected.

Vortex Valve Control Pressure

The control pressures to the vortex valve are defined as linear functions of the flapper displacement with no dynamics. Thus,

$$\Delta C_1 = \frac{\partial C_1}{\partial X} \Delta X \quad (3)$$

$$\Delta C_2 = \frac{\partial C_2}{\partial X} \Delta X \quad (4)$$

The differential control pressure is

$$\Delta C = \Delta C_1 - \Delta C_2 = \left(\frac{\partial C_1}{\partial X} + \frac{\partial C_2}{\partial X} \right) \Delta X \quad (5)$$

Load Pressure

The location of restrictors in the valve output lines as shown in Figure 16 results in the vortex valves controlling pressures to the load orifices. The compressibility time constants for the volumes between the valve and load orifices are negligible. There are, therefore, no dynamics associated with the transfer function from control to vortex valve output pressures. Thus the equations for the load pressures are

$$\Delta P_1 = \frac{\partial P_1}{\partial C_1} \Delta C_1 = \frac{\partial P_1}{\partial X} \Delta X \quad (6)$$

$$\Delta P_2 = \frac{\partial P_2}{\partial C_2} \Delta C_2 = - \frac{\partial P_2}{\partial X} \Delta X \quad (7)$$

Load Orifice Flow

The load orifices control flow to the actuator volumes. The flow equations are

$$\Delta Q_1 = \frac{\partial Q_1}{\partial P_1} \Delta P_1 - \frac{\partial Q_1}{\partial P_3} \Delta P_3 = \frac{V_1}{B} S \Delta P_3 + A_1 S \Delta y \quad (8)$$

$$\Delta Q_2 = \frac{\partial Q_2}{\partial P_2} \Delta P_2 - \frac{\partial Q_2}{\partial P_4} \Delta P_4 = \frac{V_2}{B} S \Delta P_4 - A_2 S \Delta y \quad (9)$$

Actuator Force

The actuator force relationship is defined as

$$A_1 \Delta P_3 - A_2 \Delta P_4 = (M S^2 + D S) \Delta y \quad (10)$$

Position Feedback

The actuator motion feedback is

$$\Delta Z = K_y \Delta y \quad (11)$$

Valve Transfer Function

The transfer function of the valve from torque input to vortex valve pressure output is derived by combining equations (1) through (7); thus

$$\frac{\Delta P_2}{\Delta T_\epsilon} = \frac{-r_N \frac{\partial P_2}{\partial X}}{K_\theta \left[1 - \frac{r_N^2 A_N}{K_\theta} \left(\frac{\partial C_1}{\partial X} + \frac{\partial C_2}{\partial X} \right) \right]} \quad (12)$$

where the torque error is

$$\Delta T_\epsilon = r_B A_B \Delta CP - r_Z K_Z \Delta Z \quad (13)$$

Valve-Actuator Force Transfer Functions

The transfer function from valve output pressures to actuator force is derived from equations (8) and (9) as follows:

$$\Delta P_3 = \frac{\left(\frac{\partial Q_1}{\partial P_1} \Delta P_1 - A_1 S \Delta y \right)}{\left(\frac{V_1}{B} S + \frac{\partial Q_1}{\partial P_3} \right)} \quad (14)$$

$$\Delta P_4 = \frac{\left(\frac{\partial Q_2}{\partial P_2} \Delta P_2 + A_2 S \Delta y \right)}{\left(\frac{V_2}{B} S + \frac{\partial Q_2}{\partial P_4} \right)} \quad (15)$$

The substitutions $A_1 = A_2/2$ and $V_1 = V_2/2$ are made due to actuator design. Also, since by design the load orifices are made so that their area ratio $a_2/a_1 = 2$, then

$$\frac{\partial Q_1}{\partial P_1} = \frac{1}{2} \frac{\partial Q_2}{\partial P_2} \quad \text{and} \quad \frac{\partial Q_1}{\partial P_3} = \frac{1}{2} \frac{\partial Q_2}{\partial P_4}$$

Another valid substitution is that $\frac{\partial P_1}{\partial X} = -\frac{1}{2} \frac{\partial P_2}{\partial X}$; hence, $\Delta P_1 = -\frac{1}{2} \Delta P_2$.

Applying these substitutions to equation (14) expresses it in terms of parameters related to the A_2 side of the actuator as follows

$$\Delta P_3 = \frac{\left(-\frac{1}{2} \frac{\partial Q_2}{\partial P_2} \Delta P_2 - A_2 S \Delta y \right)}{\frac{V_2}{B} S + \frac{\partial Q_2}{\partial P_4}} \quad (16)$$

The actuator force is

$$\Delta F = A_1 \Delta P_3 - A_2 \Delta P_4 \quad (17)$$

Substituting equations (15) and (16) yields

$$\Delta F = - \left(\frac{\partial Q_2}{\partial P_2} \Delta P_2 + 1.2 A_2 S \Delta y \right) 1.25 \frac{\frac{A_2}{\partial P_4} \left(\frac{\partial P_4}{\partial Q_2} \right)}{\frac{\partial P_4}{\partial Q_2} \frac{V_2}{B} S + 1} \quad (18)$$

The transfer function from load flow to actuator force output is

$$\frac{\Delta F}{\Delta Q} = - 1.25 \frac{\frac{A_2}{\partial P_4} \left(\frac{\partial P_4}{\partial Q_2} \right)}{\left(\frac{\partial P_4}{\partial Q_2} \right) \frac{V_2}{B} S + 1} \quad (19)$$

$$\text{where } \Delta Q = \frac{\partial Q_2}{\partial P_2} \Delta P_2 + 1.2 A_2 S \Delta y \quad (20)$$

The block diagram representation of the valve-actuator transfer functions developed is shown in Figure 18.

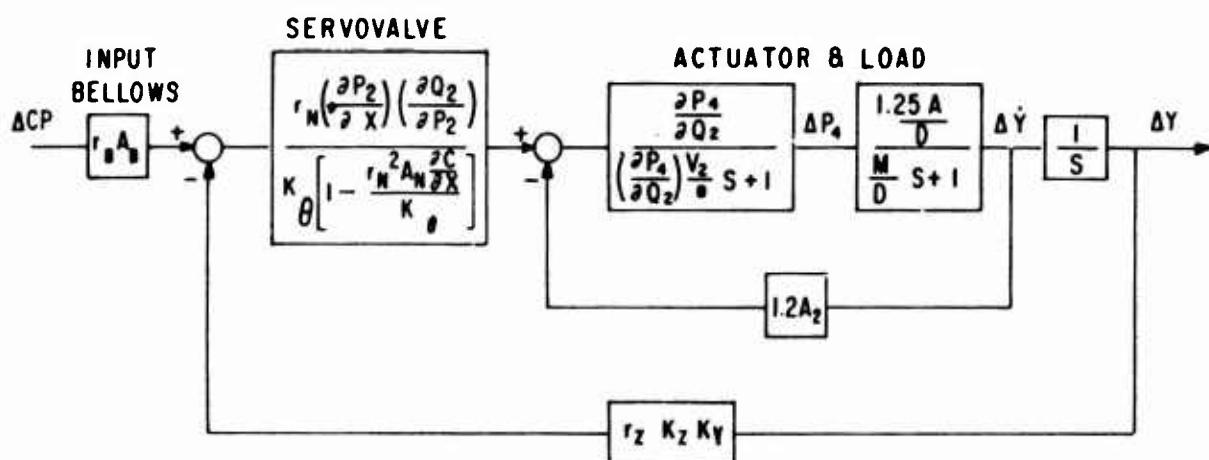


Figure 18. Fluidic Servoactuator Block Diagram of Transfer Functions.

DESIGN VALUES OF PARAMETERS

The numerical values of the parameters shown in the transfer function that were determined to meet the static gain, torque motor capacitance, and stall force requirements are:

$$r_B = 0.568 \text{ in.}$$

$$r_N = 0.52 \text{ in.}$$

$$r_Z = 1.673 \text{ in.}$$

$$A_B = 0.286 \text{ in}^2.$$

$$A_a = 0.4 \text{ in}^2.$$

$$A_N = 3.14 \times 10^{-4} \text{ in}^2.$$

$$V_a = 0.2 \text{ in}^3.$$

$$M = 1.87 \text{ lb-sec}^2/\text{ft}$$

$$K_y = 0.133 \text{ in./in.}$$

$$K_Z = 6.45 \text{ lb/in.}$$

$$K_B = 0.34 \text{ lb/in.}$$

SYSTEM GRADIENTS

The system gradient requirements and their typical values established by computation and empirically from prior test data are as follows. The system null conditions used are assumed to be:

$$P_2 = 375 \text{ psi}$$

$$P_4 = 375 \text{ psi}$$

$$C_2 = 475 \text{ psi}$$

The flapper nozzle pressure gradients are computed and also checked by tests. The typical values established for a 0.020-inch-diameter nozzle operating at the design conditions are:

$$\frac{\partial C_1}{\partial X} = 1.25 \times 10^5 \text{ psi/in.}$$

$$\frac{\partial C_2}{\partial X} = 3.13 \times 10^5 \text{ psi/in.}$$

$$\frac{\partial C}{\partial X} = \frac{\partial C_1}{\partial X} + \frac{\partial C_2}{\partial X} = 4.38 \times 10^5 \text{ psi/in.}$$

The viscous friction coefficient for the servoactuator is estimated from prior performance data. The estimated value is $D = 2.5 \text{ lb/in. / sec.}$

The static position gain defined as $\frac{r_B A_B}{r_Z K_Z K_y}$ must be equal to 0.112 in. /psi.

The physical design characteristics satisfy this requirement.

In order to meet the required loop crossover at approximately 10 Hz, the position loop gain must be 62.8; i.e.,

$$G_H = \frac{r_N}{K_\theta} \frac{\left(\frac{\partial P_a}{\partial X} \right) \left(\frac{\partial Q_a}{\partial P_a} \right)}{\left[1 - r_N^2 \frac{A_N \frac{\partial C}{\partial X}}{K_\theta} \right]} \frac{(r_Z K_Z K_y)}{(1.2 A_\theta)} = 62.8$$

Since the feedback gain ($r_Z K_Z K_y$) was established previously as 1.433 in. - lb/in. and the piston areas are established by stall force requirements, the

parameters in the valve are the only ones remaining to be adjusted to provide the desired loop gain. The valve gain must be greater than 21 cis/in-lb to satisfy the loop gain requirements.

The minimum net torsional spring rate of the flapper assembly is established by the stability requirement that the term

$$\frac{r_N^2 A_N \frac{\partial C}{\partial X}}{K_\theta}$$

should not exceed 0.7. For this condition and for the design parameters selected, K_θ must be greater than 53 in.-lb/rad.

The blocked load force gain of the valve-actuator establishes the threshold capability of the servoactuator. This gain is defined as:

$$G_F = 1.25 \frac{r_N}{K_\theta} \left(\frac{\partial P_a}{\partial X} \right) \left[1 - \frac{r_N^2 A_N \frac{\partial C}{\partial X}}{K_\theta} \right]$$

Assuming a breakout friction force of 15 lb, the force gain required to provide the breakout force with 0.04 psi (1%) input signal is 375 lb/psi. This indicates that the valve pressure gain $\partial P_a / \partial X$ required to be compatible with the other selected design value is 14×10^4 psi/in. The flapper nozzle-vortex valve combination is designed to meet this requirement. Test data on the original vortex valve servoactuator show values in the order of 9×10^4 psi/in.

Restrictors are located in the load lines to establish the desired valve flow gain without sacrificing the high static pressure gain desired. In addition, the restrictor increases the inner flow loop gain so that the inner loop response is increased. It, in effect, provides the system damping necessary for stable loop crossover with the specified inertia loading. The load orifice resistance must be selected to provide an inner loop gain greater than 10, as well as to provide desired valve gain. The resistance required to provide the valve gain of 21 cis/in.-lb is approximately 200 psi/cis. This gives a satisfactory inner loop gain of 19.2. The corresponding inner loop crossover frequency is 320 rad/sec.

The servoactuator block diagram with the numerical values of the transfer functions is shown in Figure 19. The fluid bulk modulus (B) is assumed to be 150,000 psi in the calculated results.

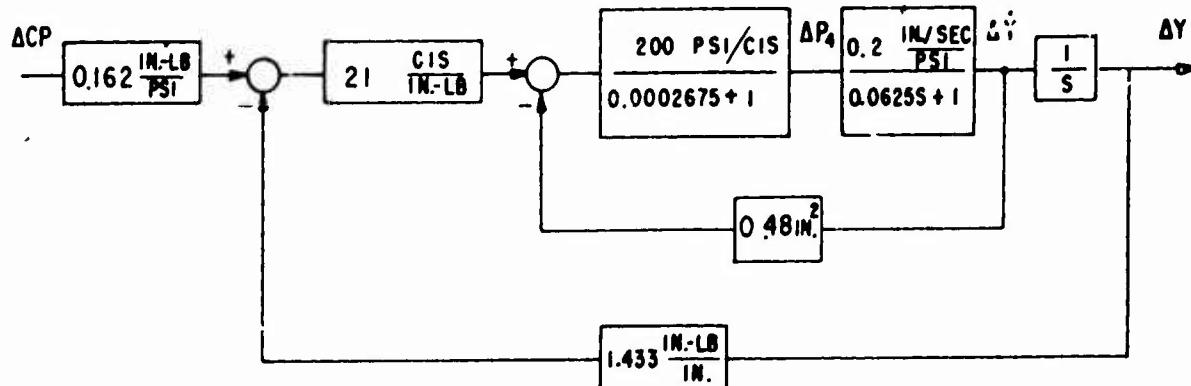


Figure 19. Fluidic Servoactuator Transfer Function.

The closed-loop frequency response for the fluidic servoactuator design represented in the block diagram is shown in Figure 20.

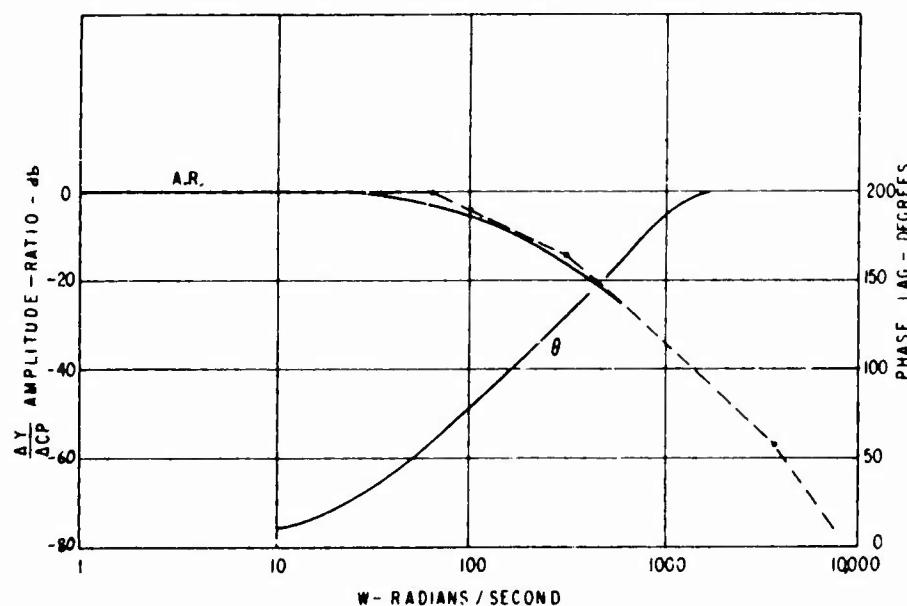


Figure 20. Fluidic Servoactuator Closed-Loop Frequency Response Calculated.

APPENDIX II

ACCEPTANCE TEST PROCEDURES

The three fluidic servoactuators shall be tested and shall successfully complete all tests required by the specifications.

Test Conditions

The following test conditions apply to all tests described under the test methods of this specification:

1. Cleaning

All oil and grease, or other corrosion-resisting compounds, shall be removed from the interior parts of the actuator before testing.

2. Test Fluid

The fluid used in the performance of all tests shall be the MIL-H-5606 hydraulic fluid. The fluid temperature shall be maintained at $100 + 20^{\circ}\text{F}$ unless otherwise noted.
 $- 10^{\circ}\text{F}$

The fluid shall be passed through a 10-micron nominal (25-micron absolute) filter before entering the unit.

3. Test Setup Environment

Unless otherwise noted:

(a) Ambient temperature shall be $25^{\circ} \pm 10^{\circ}\text{C}$ ($77^{\circ} \pm 18^{\circ}\text{F}$).

(b) Ambient pressure shall be 28 to 32 inches of mercury.

(c) Relative humidity shall be 90% or less.

4. Pressure Measurement

All pressures measured shall be gauge pressures (psig). Calibrated gauges shall be used for pressure measurement.

5. Temperature Measurement

Calibrated thermometers or thermocouples shall be used for temperature measurement.

6. Test Pressure

(a) Unless otherwise noted, the following test pressure shall apply to testing required by the Section of this specification entitled Test Methods.

(b) Supply Pressure

Supply pressure to the servoactuator shall be maintained at 1000 ±100 psig for all flows.

(c) Return Pressure

Return pressure shall be maintained at 60 ±50 psig for all flows.

Test Methods

The following tests shall be performed on each of the three fluidic servo-actuators provided under the contract containing this specification.

1. Examination of Product

Each fluidic servoactuator shall be carefully examined to determine conformance with the requirements of this specification in regard to workmanship, marking, and conformance to applicable drawings.

(a) Finish of Sliding Members

The smoothness of finish on sliding surfaces such as cylinder bores, piston heads, piston rods, etc., and other parts as specified on the manufacturing drawings, shall be determined with a profilometer, comparator, Brush Surface Analyzer, or other comparison equipment such as surface roughness comparison samples, provided the accuracy of the measuring equipment is within ±5 microinches of the required value.

(b) Physical Defect Inspection

Aluminum or aluminum alloy parts shall be anodized according to good commercial practice. There shall be no cracks or injurious defects.

(c) Piston Friction Test

Before assembling the lock-piston, measure the piston friction with the pressure and return ports open to the atmosphere. The continuous force required to move the piston through its full stroke shall not exceed 15 lb applied at the piston. Record the actual maximum value measured for each direction of travel.

(d) Stroke Measurement Test

Measure the stroke of the cylinder. The stroke shall be within the limits 0.86 to 0.94 inch.

2. Proof Pressure Test

Apply 2250 psig to the supply pressure port with the return port open to the atmosphere and perform the following:

- (a) Apply full retract signal to the fluidic force motor; the cylinder shall be fully extended. Hold for five minutes.
- (b) Apply full extend signal to the fluidic force motor; the cylinder shall be fully retracted. Hold for five minutes.
- (c) As a result of the previous steps, there shall be no external leakage, loosening, bending, or deformation of parts of the actuator.
- (d) There shall be no static external leakage; dynamic external leakage shall not exceed 1 drop over 25 cycles for any external seal.

3. Operation and Leakage Test

Each fluidic servoactuator shall be cycled through at least 50 full strokes (90% of maximum stroke) at 1 cps to demonstrate satisfactory operation, stroke adjustment, and leakage characteristics. This shall be accomplished with 1000 psig applied at the inlet port, 60 psig back pressure at the return port, and a load of 100 lb applied at the output end. Cycling shall be controlled by means of the fluidic force motor. There shall be no static leakage, and dynamic leakage is permitted at the rate of 1 drop per 25 cycles.

4. Limit and Ultimate Load Tests

Each fluidic servoactuator shall be restrained in a fixture capable of simulating all the limit and ultimate loading conditions required by this specification.

(a) Limit Load Compression Test

Shut off system pressure, allowing the locking mechanism to engage and center the piston. Apply a limit compressive load of 1500 lb to the output end with the input end restrained. Maintain the applied load for five minutes. There shall be no binding, loosening, or deformation of parts of the cylinder.

(b) Limit Load Leakage Test

Apply hydraulic system pressure to the inlet port, allowing the lock plunger to retract. Slowly cycle the cylinder through 50 full strokes by means of the servovalve to demonstrate normal operation. The output end shall be unrestrained. There shall be no static leakage, and dynamic leakage is permitted at the rate of 1 drop per 25 cycles.

(c) Limit Load Tensile Test

Repeat 4 (a) and 4 (b) above, except apply a limit tensile load of 1500 lb for five minutes.

(d) Piston Unlock Test

Each fluidic servoactuator shall be activated with balanced fluidic pressure input and no load. The supply pressure shall be increased and the supply pressure at which complete unlocking occurs shall be 250 ± 50 psig. The activation piston output transient recorded shall not exceed 0.018 inch.

(e) Piston Lock Centering Test

Each fluidic servoactuator shall be loaded with a 40-lb tensile load at the output end. The cylinder shall be extended to full length by means of the fluidic torque motor. Return pressure shall be reduced to zero and system pressure shall be shut off, allowing the spring-loaded plunger to re-center and lock the piston. The total time required to complete this actuation shall be recorded and documented and shall not exceed 1 second.

5. Threshold Motion Test

With 1000 psig applied to the supply port, and 60 psig back pressure applied to the return port, perform the following:

- (a) From the neutral signal, apply fluidic input differential pressure (to move flapper valve) gradually until motion is noted at the piston.
- (b) While carefully observing the piston of the cylinder for movement, slowly decrease the input differential pressure. Note and record the differential pressure with a pressure transducer at the instant the piston of the actuator starts to move.
- (c) While carefully observing the output rod for movement, slowly increase the input differential pressure. Note and record the reading of the pressure transducer at the instant the piston of the cylinder starts to move. The difference between this reading and the reading of the previous step [paragraph (b) above], which is the threshold, shall not exceed 0.08 psi.

6. Pressure Null Shift Test

With a balanced fluidic pressure input and no load, the supply pressures shall be varied over a range of 80% to 110% of rated pressures. The actuator motion output recorded shall be less than 0.036 inch.

7. Performance Characteristics

The dynamic gain and frequency response shall be determined and plotted for each fluidic servoactuator operating under the normal conditions defined by the Section entitled Test Methods of this specification with sinusoidal input signal equivalent to $\pm 25\%$ of rated signal and mass and friction loads specified in paragraph 3f(5) of the fluidic servoactuator design specification.

8. Low-Temperature Performance Test

Low-temperature tests with fluid temperature down to 40°F shall be conducted on one servoactuator to establish null shift, gain, and response changes.

APPENDIX III

TEST FACILITIES

Development and acceptance testing of the servoactuators were conducted on a hydraulic test stand. MIL-H-5606 oil at pressures up to 3000 psig was available. A sketch of the test setup is shown in Figure 21. The actual test installation is shown in the photograph in Figure 22. The test setup is designed for dynamic and static evaluation of the servoactuators under simulated load conditions.

For dynamic testing, the inertia load could be varied from 0.2 to 2.0 slugs by changing weights and radius of gyration on the load simulator. The friction load of 5 lb is simulated by the friction in the test setup bearings and in the loading piston and cylinder.

Tensile loading (up to 40 lb) during piston centerlock testing was applied by regulating air pressure to the loading piston-cylinder. This load was applied on the simulator at a link ratio of 3.5. For the loading piston areas of 1.463 in.² in tension and 1.77 in.² in compression, a differential pressure of 8 and 6.4 psi provided a loading of 40 lb at the fluidic servoactuator.

The limit tensile and compressive loads were applied by means of a screw-jack and force gauge arrangement.

The fluidic signal (ΔCP) of ± 4 psid at a quiescent level of 1306 psig was generated by an electrohydraulic driver consisting of an electromagnetic torque-motor-driven, single-stage flapper nozzle valve. Pressure feedback was provided around the signal generator to improve its stability and null holding capability. The output range of the signal generator was limited to less than 10 psid by means of a 0.040-inch-diameter orifice load across the output lines. This orifice in effect made the signal generator impedance more representative of that in the fluidic SAS.

An Unholg-Dickie vibration machine, Model No. 91A, was used for vibration testing in accordance with MIL-STD-810B, Helicopter Test Curve B.

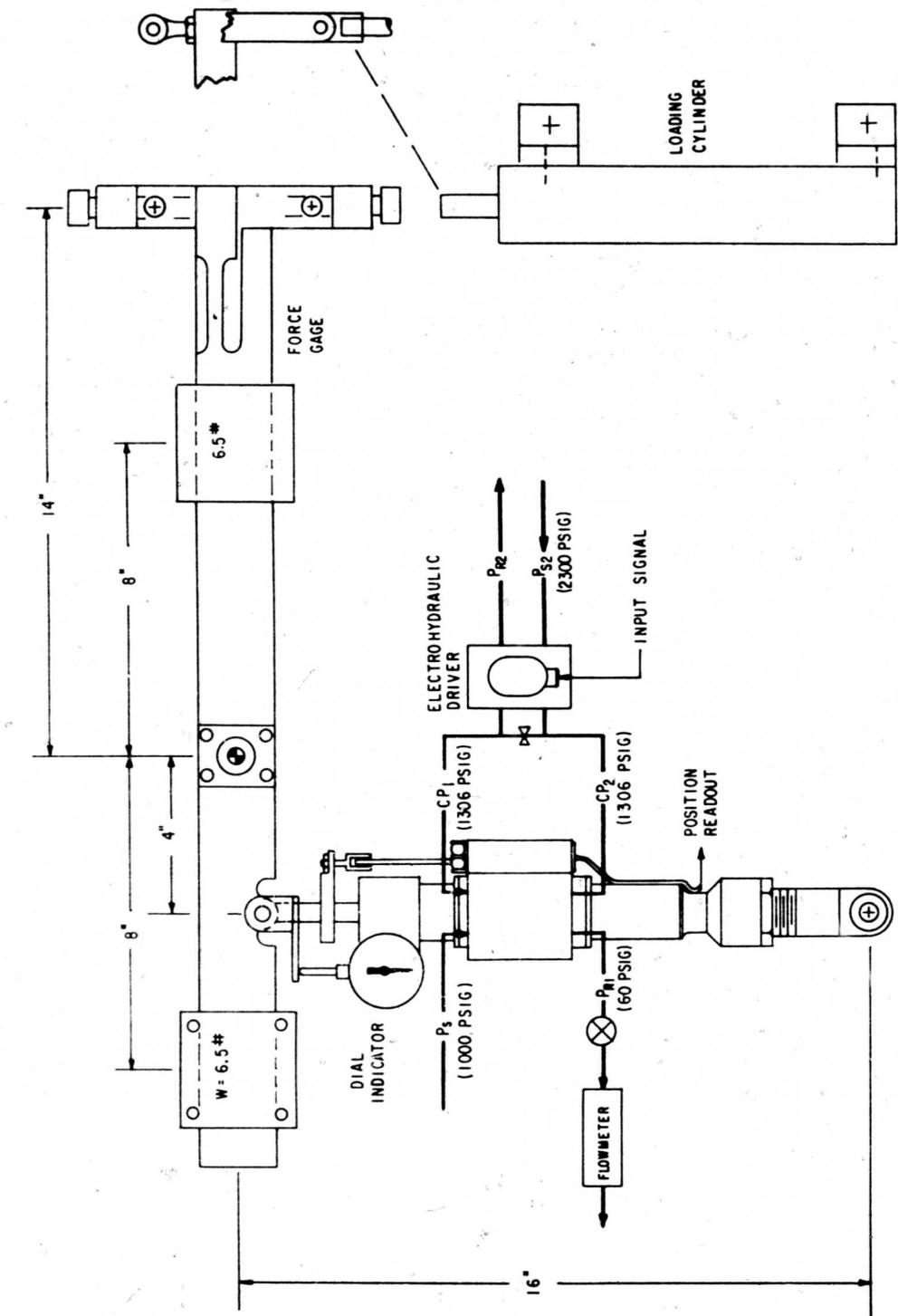


Figure 21. Fluidic Servoactuator and Load Simulator Installation.

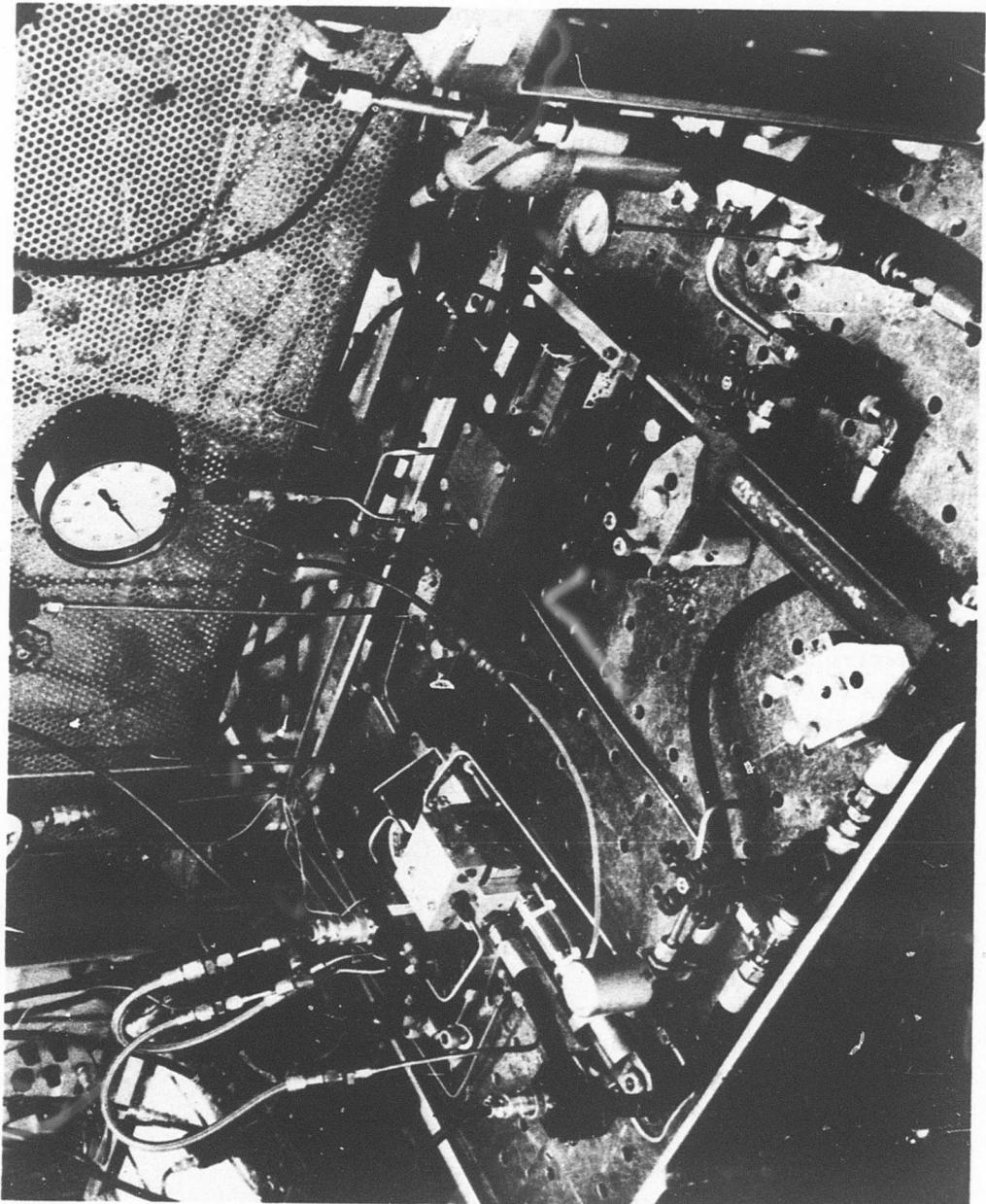


Figure 22. Fluidic Servoactuator Test Installation.

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13. ABSTRACT This report covers the developmental work accomplished on a program to furnish three fluidic servoactuators for a three-axis fluidic stability augmentation system (FSAS). The servoactuators, designed to the specification for a helicopter flight control system, are classified as "experimental" but were designed to meet safety-of-flight requirements since they were intended for flight test in a hydraulic SAS. A servoactuator controlled by a two-stage fluidic servovalve was analyzed and designed. The servoactuator hardware was fabricated and bench tested. The servoactuator bench testing, using a load simulator, showed that the fluidic servoactuator meets all of the basic acceptance test requirements. The salient features are maximum force output of 100 lb, position frequency response flat to 7 Hz when operating with MIL-H-5606 oil supplied at 1000 psig, and a quiescent flow of 0.6 gpm.		

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